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13. ABSTRACT *(Maximum 200 words)*

The goal of this project is to develop novel configurations of heat exchangers and stacks for thermoacoustic heat engines. The approach will be to use anisotropic systems, such as made possible by glass or plastic capillary array technology. A part of the project involves the development of high power drives and acoustic resonators for testing the new systems. Patent disclosures have been prepared for a stack/heat-exchanger configuration and a high power drive.

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ANISOTROPIC HEAT EXCHANGERS/STACK CONFIGURATIONS FOR THERMOACOUSTIC HEAT ENGINES

This annual summary report presents the accomplishments for ONR grant N00014-93-1-1127,"Anisotropic heat-exchangers/stack configurations for thermoacoustic heat engines". The goals are a) to use glass or plastic capillary array technology for the development of composite, anisotropic stack/heat-exchanger systems, and b) to develop high power drives and shock-avoiding acoustic resonators for testing the new systems. An important accomplishment has been the preparation of a patent disclosure for our proposed composite stack/heat-exchanger system; the disclosure is presented in an appendix.

General developments

A new graduate student, David Zhang, has been working on the thermoacoustic refrigerator project for about one year. He has been able to complete important theoretical calculations relevant to the operation of the stack/heat-exchanger system, but has had to undergo considerable training in order to build apparatus, take measurements, etc. Specific accomplishments are described below.

Theoretical Calculations for a Stack/Heat-exchanger System

The elements of our proposed anisotropic stack/heat-exchanger system are presented in the patent disclosure in the appendix. Briefly, the system uses capillaries to form a stack, but the capillaries have sections with reduced outer diameters at each end. When the capillaries are bonded together to form the stack, the reduced diameters form narrow slits through which heat-exchanger fluid may flow. The theoretical question to be addressed was whether having to push the heat-exchanger fluid through narrow slits would be effective or detrimental. After spending some time studying heat-exchanger texts, the student was able to show that the narrow slit system should work well. An outline of the calculation may be found in the appendix.

Development of a High Power Test Resonator

In order to test the new stack/heat-exchanger system at high powers, an acoustic resonator with a high amplitude drive has been constructed. The resonator is 2.5 cm in diameter and 70 cm in length. The drive is a model aircraft engine, capable of rotating at 20000 RPM (over 300 Hz), with a diameter matching the resonator and a stroke of 2.5 cm. The engine was modified so that the intake and exhaust ports are sealed. A problem was how to drive the shaft of the engine, since a powerful high-speed motor, with regulated speed, was required. After considerable searching, it was found that a readily available woodworking router was ideally suited. Such routers may have speeds variable from 8000 to 24000 RPM, with a shaft transducer and feedback system which maintains constant speed under varying loads.

The graduate student has tested the acoustic resonator and drive, and has measured a resonant frequency of 240 Hz and an acoustic pressure amplitude of 0.8 atm, consistent

with the particle velocity produced by the drive.

Current and Other Funding

Other research grants include:

1. NSF Division of Materials Research, Condensed Matter Physics Program, DMR 93-06791, 249,000/3 yr, which includes 1 man-month of the principal investigators time.

2. ONR, Physics Division, N00014-92-J-1186, November 1, 1991 to October 31, 1994, 300,000/3 yr, "Innovative acoustic techniques for studying new materials and new developments in solid state physics"; includes 1 man-month of time for the principle investigator.

3. ONR, Physics Division, N00014-93-1-0779, June 1, 1993 to May 31, 1996 105,000/3 yr, "Training students in new acoustic techniques for studies of fracture and nondestructive evaluation of exotic materials"; includes no time for the principal investigator.

APPENDIX

Invention Disclosure for a Stack/Heat-exchanger Unit for Thermoacoustic Heat Engines

A major contributor to the depletion of the earth's ozone layer is the reaction of the ozone with chlorofluorocarbons (CFC's) which are released into the earth's atmosphere from refrigerators which leak CFC's. [Note: by physics definition, refrigerators include air conditioners, etc.] In order to satisfy current and anticipated regulations governing the use of CFC's, it will be necessary to develop new types of refrigerators which do not use CFC's. A promising technology involves the thermoacoustic effect, in which the oscillatory motion of a gas in an acoustic field is coupled to a temperature gradient at a solid surface parallel to the motion. Reviews of this effect and its application in refrigerators and other heat engines (some nearing commercialization), are available in the literature.[1,2]

In order to increase the heat carrying capacity of the thermoacoustic heat engine, a large number of solid surfaces are used in a parallel configuration, as in a stack of plates, a spiraling sheet, or an array of capillaries; this part of the thermoacoustic heat engine is referred to as the "stack". An exploded view of a thermoacoustic refrigerator (TAR) is shown in Fig. 1. At the left in Fig. 1 is the acoustic driver, represented as an oscillating piston. Next is an open section of a longitudinal standing wave acoustic resonator, followed by the stack. At each end of the stack are heat exchangers, represented by fluid-carrying pipes in Fig. 1, which would connect the refrigerator to an ambient temperature reservoir and to the load to be cooled. Finally, at the right end in Fig. 1 is the end cap for the acoustic resonator. The sealed acoustic resonator is filled with a non-CFC gas, such as a helium-argon mixture. It should be noted that the spacing of the surfaces in the stack (or the inside diameter of capillaries in an array) is on the order of the thermal penetration depth for the gas, typically a few hundred microns.

Key elements in a high power thermoacoustic refrigerator are the heat exchangers at the ends of the stack, illustrated in Fig. 2. For an isothermal heat-exchanger, the length of the exchanger (in the direction of the gas particle velocity) should be on the order of the gas particle peak-to-peak displacement (as large as several millimeters). Such an isothermal exchanger poses a fundamental limitation, because the isothermal surface (unlike the surface with a temperature gradient in the stack) is a source of acoustic power loss. Another difficulty arises from the disparity in the length scales between the stack (with a scale of several hundred microns) and the heat exchanger pipes (with a scale of several mm). The TAR could be improved if the heat-exchanger were incorporated into the stack with a matching length scale. This would form an anisotropic stack/heat-exchanger unit, which could transport large heat flows laterally (across the stack) but not longitudinally (along the stack). If such a heat-exchanger used a flowing fluid, rather than heat conduction, to transport the heat, then one could not only handle higher heat loads, but one could also have graded exchanges with the external heat exchangers. That is, the temperature of the heat-exchanger fluid entering and exiting the external heat-exchanger could be made to match the temperature at the point of entry or exit of the exchanger in the stack.

A possible anisotropic stack/heat-exchanger unit is illustrated in Fig. 3; for simplicity the unit has been drawn with a square cross section, and lengths are not drawn to scale. The unit is composed of an array of capillaries (thin-walled, a few hundred microns in diameter), as in a conventional stack of this type, but now the capillaries have a reduced outer diameter near each end, as illustrated by the single capillary in Fig. 3. In the stack/heat-exchanger unit, the capillaries are bonded together so that the regions having the full diameter are completely sealed, while the regions of reduced diameter form narrow ($\sim$100 μm) open slits. The acoustic gas oscillates inside the capillary pores, and a heat-exchanger fluid flows at right angles through the narrow slits between the capillaries. The heat to or from the acoustic gas near the ends of the stack conducts through the thin walls of the capillaries, and is transported by the heat-exchange fluid. A question which arises is: Will having to push the heat-exchange fluid through narrow slits be detrimental? To address this question, some heat-exchanger fundamentals should be reviewed.

Some basic concepts for heat-exchangers are illustrated in Fig. 4. We consider a geometry consisting of two parallel isothermal walls at temperature T_w with a heat-exchange fluid flowing between them, entering at a temperature $T_0 < T_w$. Heat leaving the plates is carried away by the moving heat-exchange fluid. Two basic modes for heat-exchange fluid flow, laminar and turbulent, are illustrated in Figs. 4a and 4b. The flow profile for laminar flow is shown by the parabola-shaped curves in Fig. 4a; the flow is parallel to the walls, having a maximum in the center and dropping to zero at the walls. At the entrance (the left end in Fig. 4a) the temperature rises abruptly at the walls. For a wall spacing typical of refrigeration tubing (several mm), the consequence of these profiles is that the heat diffuses only a small distance from the walls where the flow is small, so that little heat is carried away. If the fluid is pushed fast enough between the walls, the flow becomes turbulent, as shown in Fig. 4b. In this case, perpendicular flow of the fluid near the walls convects the heat into the center of the channel where it is more effectively carried away by the mean flow of the fluid. For typical millimeter sized refrigeration tubing, turbulent flow is better for heat exchange. We now use a theoretical calculation to see if making the walls close together (as in narrow slits) might make the laminar flow situation more effective.

The theory involves the equation of continuity, the Navier-Stokes equation, and conservation of energy, including heat conduction, for an incompressible fluid:

$$\frac{\partial \rho}{\partial t} + \vec{\nabla} \cdot (\rho \vec{v}) = 0 \tag{1}$$

$$\rho \frac{\partial \vec{v}}{\partial t} + \rho \vec{v} \cdot \vec{\nabla} \vec{v} = -\vec{\nabla} P + \eta \nabla^2 \vec{v} \tag{2}$$

$$T \frac{\partial S}{\partial t} + T \vec{v} \cdot \vec{\nabla} S = \frac{1}{\rho} \vec{\nabla} \cdot \left(\kappa \vec{\nabla} T \right) \tag{3}$$

where, for the heat-exchange fluid, ρ is the mass density, $\vec{v}$ is the flow velocity field, P is the pressure, η and ξ are the shear and bulk viscosities, κ is the thermal conductivity, T is temperature, and S is the entropy.

For steady, incompressible flow between parallel walls, with spacing $2y_0$ as illustrated in Fig. 4c, the first two equations yield:

$$P\left(x,y\right) = P + \Delta P\left(1 - \frac{x}{L}\right) \tag{4}$$

$$\vec{v}\left(x,y\right) = \frac{\Delta P y_0^2}{2\eta L}\left(1 - \frac{y^2}{y_0^2}\right) \tag{5}$$

where L is the length from the left to the right end of the heat-exchanger and ΔP is the pressure drop across L. The net flow of heat exchange fluid is proportional to y_0^2, so that it would seem that narrow slits would be a disadvantage. Indeed, this impedance limits the flow to laminar for reasonable values of ΔP. However, the calculation must be continued to find the the net heat transport, given by the thermal conductance:

$$\frac{\dot{Q}}{\Delta T} = \left(\frac{\Delta P \rho C_p}{8\eta L}\right)\sum_{n=1}^{\infty} A_n y_0^2\left[1 - e^{-(\zeta_n/y_0)^4}\right] \tag{6}$$

where

$$\zeta_n = \left[\frac{2\gamma_n \eta L^2 \kappa}{\Delta P \rho C_p}\right]^{1/4} \tag{7}$$

$\Delta T = T_w - T_0$, and γ_n is an eigenvalue.

For large y_0 (3 mm), $\dot{Q}/\Delta T$ decreases with y_0 , reiterating that laminar flow is undesirable. However, for small y_0 it increases. For $L \sim 5$ - 10 cm, the maximum thermal conductance occurs for a wall spacing of a hundred microns, the same as the typical spacing for a stack. This favorable result may be seen as arising from the similarity between the formula for the thermal penetration depth

$$\delta = \sqrt{\frac{\kappa}{\rho C_p f}} \tag{8}$$

and that for the optimum plate spacing

$$2y_0 = \sqrt{\frac{\kappa}{\rho C_p\left(v/L\right)}} \tag{9}$$

where f is the frequency of the acoustic resonance and v is the average heat-exchanger fluid flow velocity. For a typical TAR, f and v/L are comparable.

Another quantity to consider is the energy lost through viscosity in pushing the heat-exchange fluid through the narrow slits. This is calculated with the product of ΔP and v. If one uses typical values for the parameters for a stack, one finds that the performance of a stack/heat-exchanger unit as illustrated in Fig. 3, with 100 μm slits, is comparable to that of a conventional TAR heat exchanger formed with 3 mm ID copper refrigerator tubing, but with the advantages that there is much better thermal contact with the thermoacoustic gas in the capillaries, and by using several sections of slits, a graded heat exchanger may be formed. Furthermore, a glass or plastic capillary array incorporated into the heat-exchanger or stack has the advantages of being mechanically (and chemically) robust, easily cleaned, and having high porosities.

References

1. G. W. Swift, J. Acoust. Soc. Am. **84**, 1145 (1988). Thermoacoustic engines.

2. G. W. Swift, *Physics Today* , July 1995, p. 22.

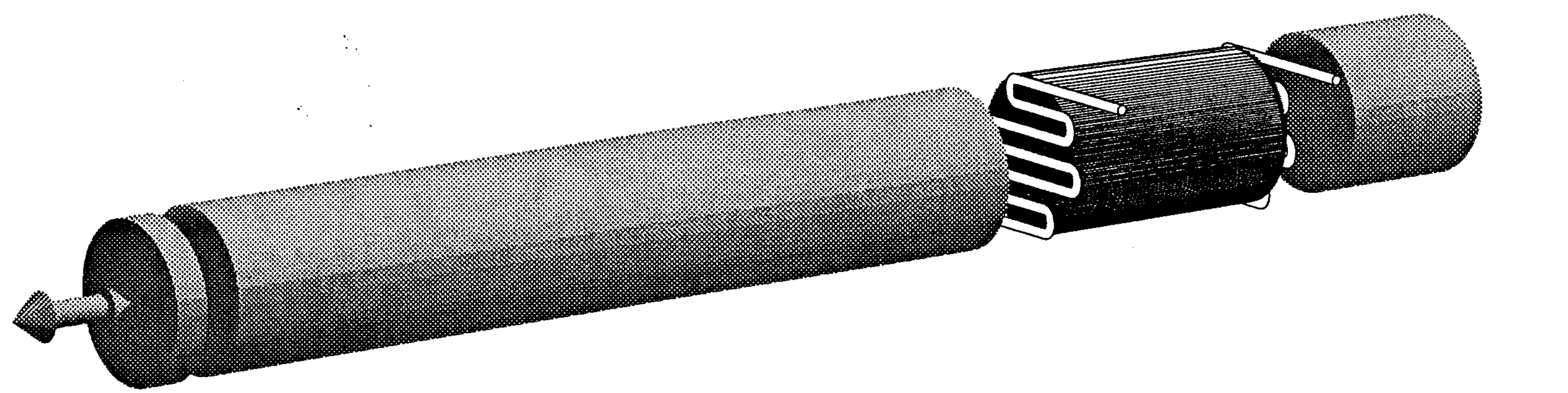


Fig. 1

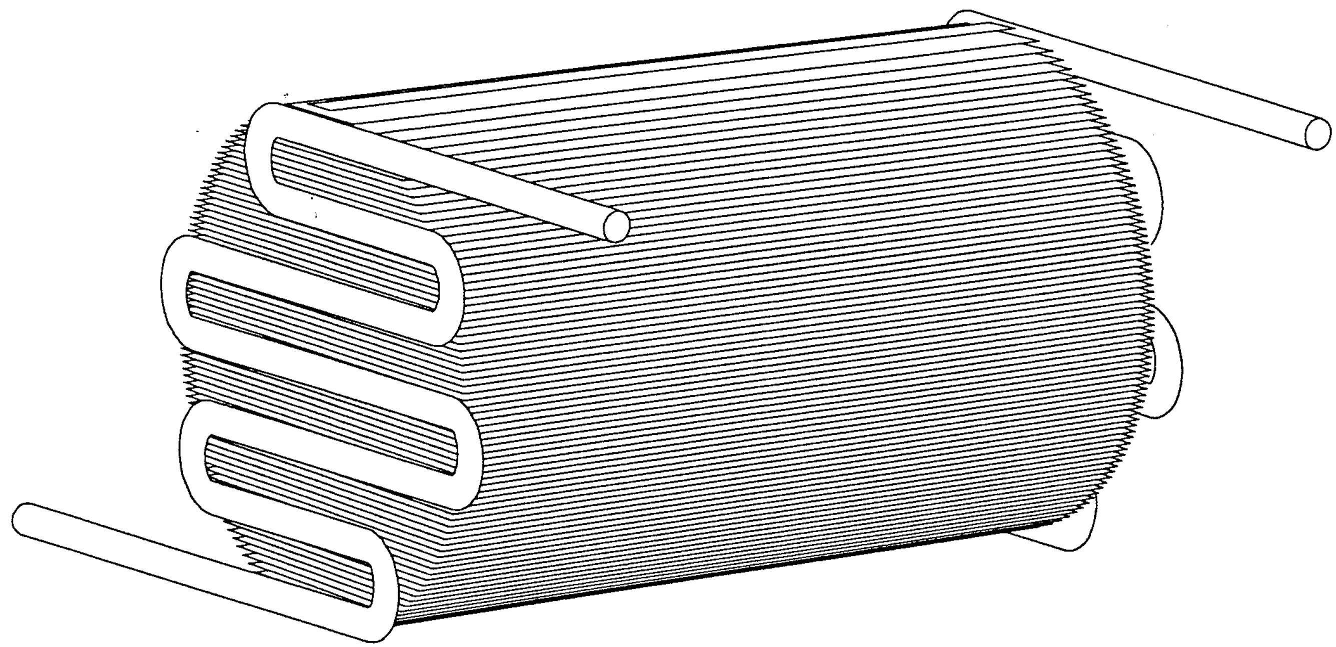


Fig. 2

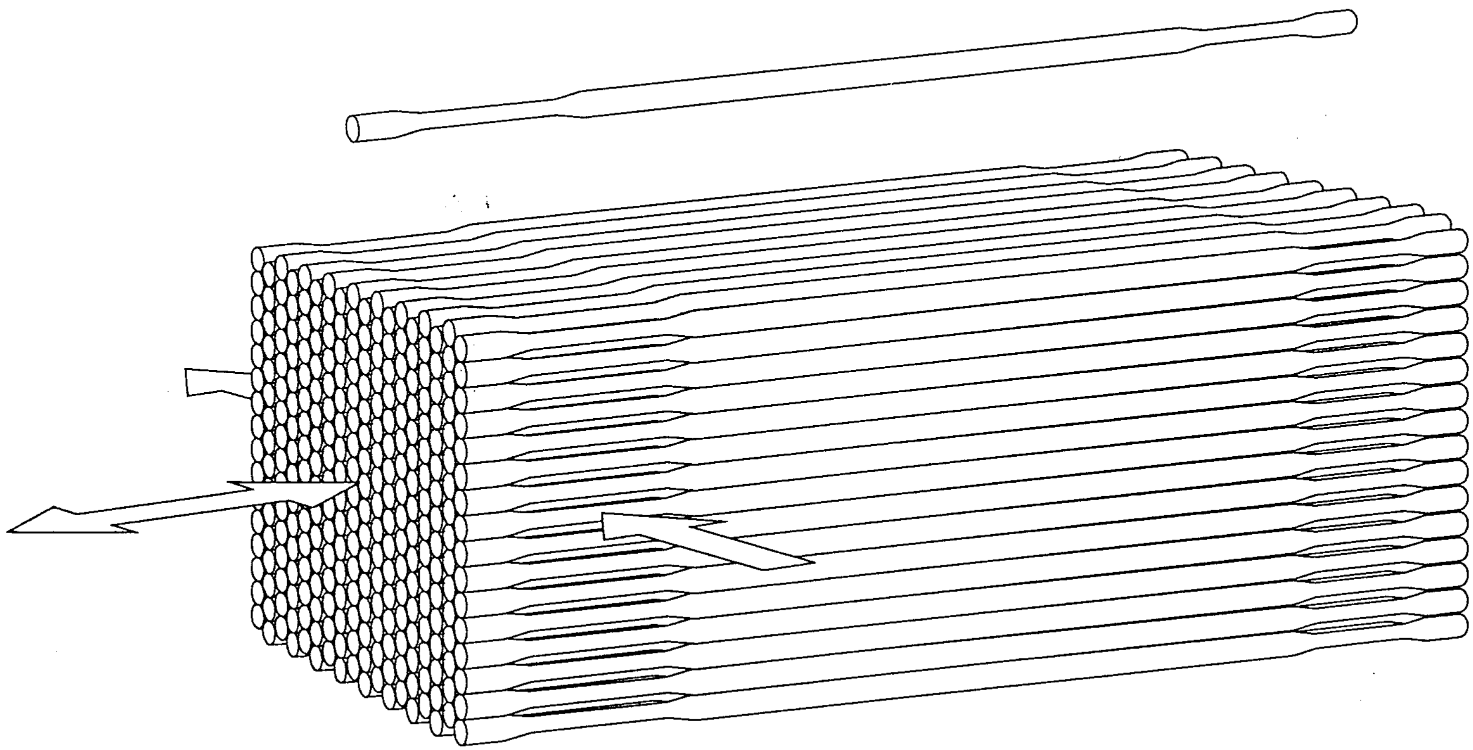
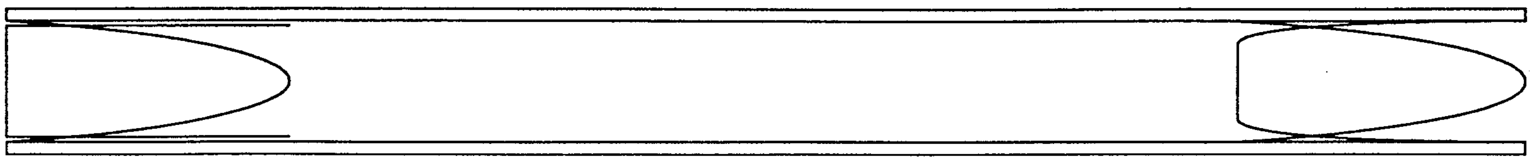


Fig. 3

(a)



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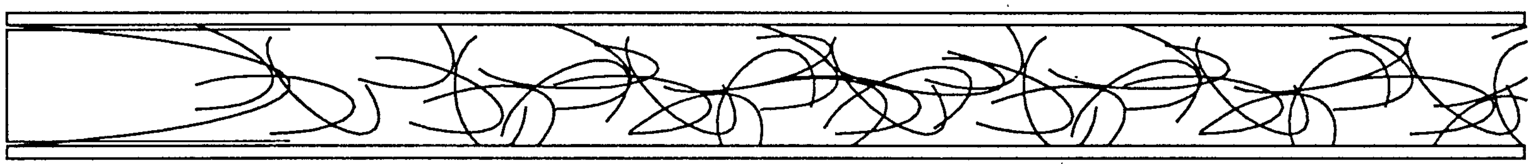

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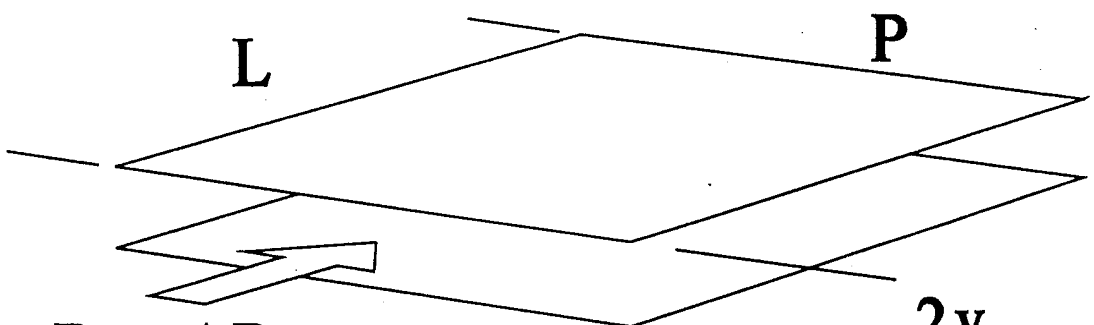

Fig. 4